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FRICTION COEFFICIENTS IN A VANELESS DIFFUSER

By W. Byron Brown

Flight Propulsion Research Laboratory
Cleveland, Ohio

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SUMMARY

Friction coefficients for three constant-area vaneless diffusers, used in conjunction with a centrifugal impeller, were determined from static- and total-pressure surveys taken at several radii and from the usual over-all measurements of temperature, pressure, and air flow. The average value of the friction coefficient through the entire diffuser was approximately 50 percent higher than that for fully developed turbulent flow in smooth pipes. Friction coefficients at the diffuser entrance were about three times the pipe values. In the middle of the diffuser, the friction coefficients agreed well with smooth-pipe values.

INTRODUCTION

The conversion of kinetic energy into pressure rise in a diffuser is accompanied by significant losses. For certain applications - for example, in the design of diffusers - these losses are usually expressed in terms of a friction coefficient that is a measure of the loss in total pressure through the diffuser.

The available data on friction coefficients in vaneless diffusers are meager and show a rather wide scatter, partly because of the difficulties inherent in the determination of such coefficients and partly because of diversity in their definition. Writers differ by a factor of 2 or 4. McAdams' definition (reference 1, p. 119) is used in this paper. Skubatchevsky (reference 2) gives values for the friction coefficient in conical diffusers ranging from 0.032 to 0.100; these values were found by the calculation of the kinetic-energy losses between successive sections of the diffuser. These diffusers were straight and of circular, square, and flat-rectangular cross section. Two coefficients were suggested in 1941 by H. W. Emmons and co-workers of Pratt & Whitney Aircraft as a measure of diffuser losses; one coefficient was based on angular momentum and the other, which is the same as the one considered herein, was determined from the increase in entropy of the fluid. The entropy change was found from total- and static-pressure surveys in the diffuser. The data gave "entropy" coefficients that covered an approximate range of 0.001 to 0.032.

Because the available data on diffuser friction coefficients are scarce and have a rather wide scatter and because more accurate values would be useful in diffuser design, additional information was considered desirable. In the present investigation, static and total pressures were measured at several stations along the radius of three constant-area vaneless diffusers, and these data were used to determine velocity and pressure changes. The friction coefficient was then calculated from these changes and the hydraulic diameter. The investigation was conducted with a centrifugal impeller of conventional design, and the three constant-area diffusers, formed by two successive diameter reductions of the original diffuser.

PROCEDURE AND APPARATUS

The experimental work was done in a variable-component compressor in accordance with the recommendations of reference 3, with the exception that one radial outlet pipe was used instead of two. Data were obtained at actual impeller tip speeds of 800, 1000, and 1200 feet per second. At speeds of 800 and 1000 feet per second, the discharge pressure was held at 10 inches of mercury above atmospheric. At a speed of 1200 feet per second, power limitations prevented a complete load range at this discharge pressure; therefore each run at this speed was begun near surge and the load was increased until the power limit was reached. All runs were made at ambient-air temperature.

A conventional centrifugal impeller of 12.33-inch diameter was used with three vaneless diffusers in which the area perpendicular to the radius had the constant value of 37.4 square inches. The three diffusers were formed by successively cutting down the original 27-inch-diameter diffuser to a 24-inch and then a 20-inch diameter. The first one-half inch of the diffuser wall was faired to make a smooth front shroud.

The over-all measurements were made with standard instruments at the locations recommended in reference 4. For the pressure surveys in the diffuser, static pressures were measured at points opposite each other on the front and rear diffuser walls, and the arithmetic mean of the two readings was taken as the static pressure at that location. Inasmuch as the curvature of the diffuser profile is small, this procedure should give a satisfactory value for the static pressure at the point. The static-pressure taps were 0.020 inch in diameter and their radial location, as well as that of the total-pressure tubes, is shown in figure 1. The total-pressure tubes were 0.094 inch in diameter with one end plugged and rounded off and a hole 0.020 inch in diameter drilled in the side near the plugged end.

A total-pressure reading was taken by rotating the tube about its axis until a maximum reading was obtained. For each run, readings were taken with each tube at four stations that were one-eighth, three-eighths, five-eighths, and seven-eighths of the total distance from the front to the rear diffuser walls. The total pressure used in the calculations is the arithmetic mean of these four readings.

SYMBOLS

The following symbols are used in the report:

b	passage width
c_p	specific heat of air at constant pressure
D	impeller diameter
D_h	hydraulic diameter
f	friction coefficient
g	acceleration of gravity
H	work done on air per unit mass
L	length of flow path of air
M	moment of momentum of air
N	rotational speed
P	total pressure
p	static pressure
Q	volume flow of air at impeller outlet
R	gas constant
Re	Reynolds number
r	distance from axis of rotation
T	total temperature

t static temperature

U resultant air velocity

U_r radial component of air velocity

U_t tangential component of air velocity

V impeller tip speed

W weight flow of air

α angle between absolute velocity and tangential component

γ ratio of specific heat at constant pressure to specific heat at constant volume

ΔU_t windage loss expressed as a decrease in U_t

μ viscosity of air

ρ weight density of air

The subscript o indicates the diffuser outlet.

THEORY

Derivation of Equation for Friction Coefficient

If the energy losses in a vaneless diffuser are combined into a single term and considered as a loss in pressure due to friction, an extended form of Bernoulli's equation may be used for the determination of friction coefficients. The equation (given in reference 1, p. 127, in slightly different notation) is

$$\frac{dp}{\rho} + \frac{UdU}{g} + \frac{4fU^2dL}{2gD_h} = 0 \qquad (1)$$

where the hydraulic diameter D_h is defined as

$$D_h = \frac{4 \times \text{cross-sectional area}}{\text{wetted perimeter}}$$

McAdams states that the pressure drop due to friction in rectangular ducts may be calculated as for circular pipes, by using an equivalent diameter equal to four times the hydraulic radius (reference 1, p. 123). If the vaneless diffuser is considered as a passage of large aspect ratio, the application of this equation seems appropriate for the determination of diffuser friction coefficients.

At any radius r in the diffuser, the cross-sectional area normal to the flow path is $2\pi rb \sin \alpha$ and the wetted perimeter $2(2\pi r) \sin \alpha$. Therefore,

$$D_h = \frac{4\ (2\pi rb \sin \alpha)}{4\pi r \sin \alpha} = 2b$$

Also, for flow through the diffuser,

$$dL = dr \csc \alpha$$

and from the general gas equation

$$\rho = \frac{p}{Rt}$$

Substitution of the foregoing values for D_h, dL, and ρ in equation (1) and solving for f gives

$$f = \frac{-\frac{U}{Rgt}\frac{dU}{dr} - \frac{1}{p}\frac{dp}{dr}}{\frac{U^2 \csc \alpha}{Rgb\ t}} \tag{2}$$

Equation (2) can be modified to give values of f in terms of total instead of static pressures.

Friction Coefficients from Static Pressures

The value of $(1/p)\ (dp/dr)$ used in equation (2), is found from the curves for the static-pressure surveys. Figure 2 shows a typical curve. The velocity U is found for any radius r from its components U_r and U_t. The value of U_r is determined from the continuity equation, and that of the U_t from the equation for angular momentum with no heat transfer and no recirculation assumed.

At the impeller outlet, the tangential velocity corrected for windage losses is (reference 5)

$$U_t = \frac{gH}{V} - \Delta U_t$$

which becomes in dimensionless form

$$\frac{U_t}{V} = \frac{gH}{V^2} - \frac{\Delta U_t}{V}$$

For the impeller used in this study,

$$\frac{\Delta U_t}{V} = \frac{23.9}{10^4} \frac{ND^3}{Q}$$

where Q is measured at the impeller outlet. Thus, for this impeller,

$$\frac{U_t}{V} = \frac{gH}{V^2} - \frac{23.9}{10^4} \frac{ND^3}{Q} \tag{3}$$

With this value as a datum, U_t at any other point may be found from the relation derived in reference 6, which in the notation of this paper is

$$\frac{dM}{M} = -f \frac{\csc \alpha}{b} dr \tag{4}$$

where

$$M = rU_t$$

Calculation of the friction coefficient by equation (2) for a given set of experimental data and calculated values of U obtained by assuming widely different values of f in equation (4) demonstrated that the value of f from equation (2) was quite insensitive to changes in the value of f assumed for equation (4). The value given by the von Kármán equation (reference 1, p. 119),

$$\frac{1}{\sqrt{f}} = 4 \log_{10} (\text{Re}\sqrt{f}) - 0.40$$

was therefore considered sufficiently accurate for use in equation (4).

The Reynolds number is expressed as

$$\text{Re} = \frac{UD_h\rho}{\mu}$$

Because $U = U_r/\sin \alpha$, $W = \rho U_r \, 2\pi rb$, and $D_h = 2b$, the Reynolds number may also be written

$$Re = \frac{W}{\mu\pi r \sin \alpha}$$

A step-by-step procedure using radial increments of one-half inch was employed in finding the values of U_t and U_r. Inasmuch as the change in the quantity $(\csc \alpha)/b$ of equation (4) is small through the diffuser, the value at the diffuser entrance was used for the first increment of radius in a point-to-point solution. Then, when U_t and U_r were found for this radius, the value of $(\csc \alpha)/b$ was found for the end of the increment and used in the next step.

Substitution of the value of ρ from the general gas equation

$$\rho = \frac{p}{Rt}$$

in the continuity equation

$$W = \rho 2\pi rbU_r$$

yields

$$W = \frac{p2\pi rbU_r}{Rt}$$

or

$$U_r = \frac{WRt}{2p\pi rb} \tag{5}$$

where

$$t = T_o - \frac{U_t^{\,2} + U_r^{\,2}}{2gc_p} \tag{6}$$

Between the impeller and the outlet measuring station, T_o is assumed constant.

Friction Coefficients from Total Pressures

In addition to the static-pressure surveys that were taken for all runs, total-pressure surveys were taken for several runs. Inasmuch as these total-pressure readings provided independent data for the calculation of friction factors, they were used as a check on the results calculated from static pressure.

The relation between total and static pressures may be written

$$\frac{p}{P} = \left(1 - \frac{U^2}{2gc_pT_o}\right)^{\frac{\gamma}{\gamma-1}} \tag{7}$$

If the value of T_o (equation (6)) is substituted in equation (7) and the result is logarithmically differentiated,

$$\frac{dp}{p} = \frac{dP}{P} - \frac{\gamma}{\gamma-1}\frac{UdU}{gc_pt}$$

When c_p is replaced by $\gamma R/(\gamma-1)$

$$\frac{dp}{p} = \frac{dP}{P} - \frac{UdU}{gRt}$$

When this value of dp/p is substituted in equation (2)

$$f = -\frac{\frac{1}{P}\frac{dP}{dr}}{\frac{U^2 \csc \alpha}{Rgbt}} \tag{8}$$

Because $U = (M \sec \alpha)/r$, equation (8) can be written

$$f = -\frac{\frac{gRt}{M^2 \sec^2 \alpha}\frac{b}{\csc \alpha}\frac{dP}{P}}{\frac{dr}{r^2}} \tag{9}$$

COMPUTATIONS

Friction coefficients were determined from the static-pressure surveys by equation (2) as follows: The term $(1/p)(dp/dr)$ was evaluated from the curve of the average values of the static pressure through the diffuser by dividing the pressure difference corresponding to the two end points of the interval under consideration by the pressure at the midpoint of the interval. The evaluation was carried out for each 1/2-inch interval (radially) between $r = 6.5$ inches and $r = r_o - 0.5$ inch.

Equations (5) and (6) may be combined to give a quadratic equation in U_r, but the solution of this equation is long and complicated. Because the same solution can be obtained more easily from equation (5) by successive approximations, this procedure was used. By use of a value of U_t given by equations (3) and (4) and with a trial value for U_r, the equation was solved for t. This value of t was then used to obtain a revised value of U_r and if necessary, the process was repeated. One or two trials were usually sufficient. For any one run, U_r changed so slowly through the diffuser that the value determined for the previous 1/2-inch interval could be used to find t for the next.

The value of U_t at the impeller exit was found from equation (3). For points in the diffuser, the value of U_t was determined (for the same 1/2-in. intervals as U_r) from the value at the impeller exit and equation (4). The angle α and resultant velocity U were then easily found.

In the evaluation of friction coefficients from total-pressure readings, equation (9) written in the form

$$-f \int_{r_1}^{r_2} \frac{dr}{r^2} = \int_{P_1}^{P_2} \left(\frac{gRt}{M^2 \sec^2 \alpha} \frac{b}{\csc \alpha} \right) \frac{dP}{P}$$

(where the subscripts 1 and 2 indicate two successive stations) was integrated between limits fixed by the location of the total-pressure tubes. Because the quantities inside the parentheses are either constant or change slowly with the radius and because each of the successive integrations covers a rather small radial distance, the error introduced by leaving these quantities outside the integration sign should be quite small. Integration gives

$$f_{1-2} = \left(\frac{gR\overline{t}}{M^2 \, \overline{\sec^2 \alpha}} \right) \left(\overline{\frac{b}{\csc \alpha}} \right) \left(\log \frac{P_1}{P_2} \right) \frac{r_1 r_2}{r_1 - r_2}$$

where the bars indicate that average values of $\sec^2 \alpha$, t, and $(b/\csc \alpha)$ were used in the numerical calculations of each of the intervals. The radius with which f_{1-2} is associated is the average of r_1 and r_2.

RESULTS AND DISCUSSION

The following table presents the radial variation in the friction coefficient computed from static-pressure data for the three diffusers:

Radius (in.) ↓ \ Diffuser diameter (in.) →	Friction coefficient		
	20[a]	24[b]	27[c]
7.0	0.0098	0.0111	0.0093
7.5	.0071	.0074	.0068
8.0	.0071	.0051	.0047
8.5	.0024	.0040	.0033
9.0	.0031	.0033	.0030
9.5		.0031	.0031
10.0		.0033	.0035
10.5		.0031	.0037
11.0		.0032	.0040
11.5			.0044
12.0			.0051
12.5			.0060

[a]Average of 22 runs.
[b]Average of 20 runs.
[c]Average of 25 runs.

For each diffuser the values of the friction coefficient at the indicated radii were calculated for all tip speeds and all loads except those in the region of flow cut-off where the data were unreliable. Values near surge were included, though their accuracy is in some doubt. In this region temperature values probably are abnormally high because of recirculation and therefore give abnormal values of U_t, and therefore of f. The values shown in the table are averages.

Average values of the friction coefficient determined from total pressures are shown in the following table for the eight runs of the 27-inch diffuser, in which total pressures were observed (no total-pressure readings were taken in the other diffusers):

Survey-station radius (in.)	Friction coefficient
6.50-7.25	0.0083
7.25-8.00	.0073
8.00-9.50	.0042
9.50-11.50	.0042
11.50-13.00	.0036

The friction coefficients calculated from total and static pressures are significantly higher at the diffuser entrance than in the intermediate region and probably reflect mixing losses and rapid readjustment of the velocity profile. For the 20-inch and 24-inch diffusers, no apparent increase exists in friction coefficient near the exit; whereas for the 27-inch, a definite increase in the static-pressure coefficient appears to exist in this region, but none in the coefficient computed from total pressures. Both of these circumstances seem to point to incipient flow separation at the exit of the 27-inch diffuser.

In the central region of the diffuser (between radii of 8.5 and 11 in.), the values for the friction coefficient computed from either total or static pressures agree with each other and with values obtained from von Karman's equation for fully established turbulent flow in smooth pipes (fig. 3).

Average values of friction coefficient based on static pressures through the entire diffuser for all the runs at a given tip speed are presented in the following table for each of the diffusers, together with the mean value for the diffuser at all tip speeds:

Average friction coeffi-cient	Actual tip speed (ft/sec)	Number of runs
Diffuser diameter, 20 in.		
0.0047	800	8
.0057	1000	8
.0059	1200	6
Mean 0.0054		Total 22
Diffuser diameter, 24 in.		
0.0052	800	7
.0051	1000	7
.0053	1200	6
Mean 0.0052		Total 20
Diffuser diameter, 27 in.		
0.0055	800	10
.0049	1000	9
.0038	1200	6
Mean 0.0049		Total 25

As might be expected, the mean values of the friction coefficient increase as the diffuser diameter decreases because the region of high friction coefficients occupies a larger percentage of the total area in the small diffusers than in the large. For the range of diameters investigated in this study, the average friction coefficient for all the diffusers was approximately 50 percent higher than the smooth-pipe values.

SUMMARY OF RESULTS

Friction coefficients calculated from total- and static-pressure surveys in the three constant-area diffusers used in conjunction with a centrifugal impeller indicated comparatively large total-pressure losses (three times pipe values) at the diffuser entrance, smaller losses near the exit, and minimum losses (approximately those for smooth-pipe flow) in the intermediate section of the diffuser. Friction coefficients computed from total pressures were significantly larger than smooth-pipe values only at the diffuser entrance. Losses in the flow in the midsection of the diffuser apparently were about the same as in smooth pipes.

Average values of the static-pressure friction coefficient over the entire flow path in the diffuser were about 50 percent higher than those found from von Kármán's equation for turbulent, smooth-pipe flow. The over-all friction coefficient increased with decreasing diameter.

Flight Propulsion Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, March 11, 1947.

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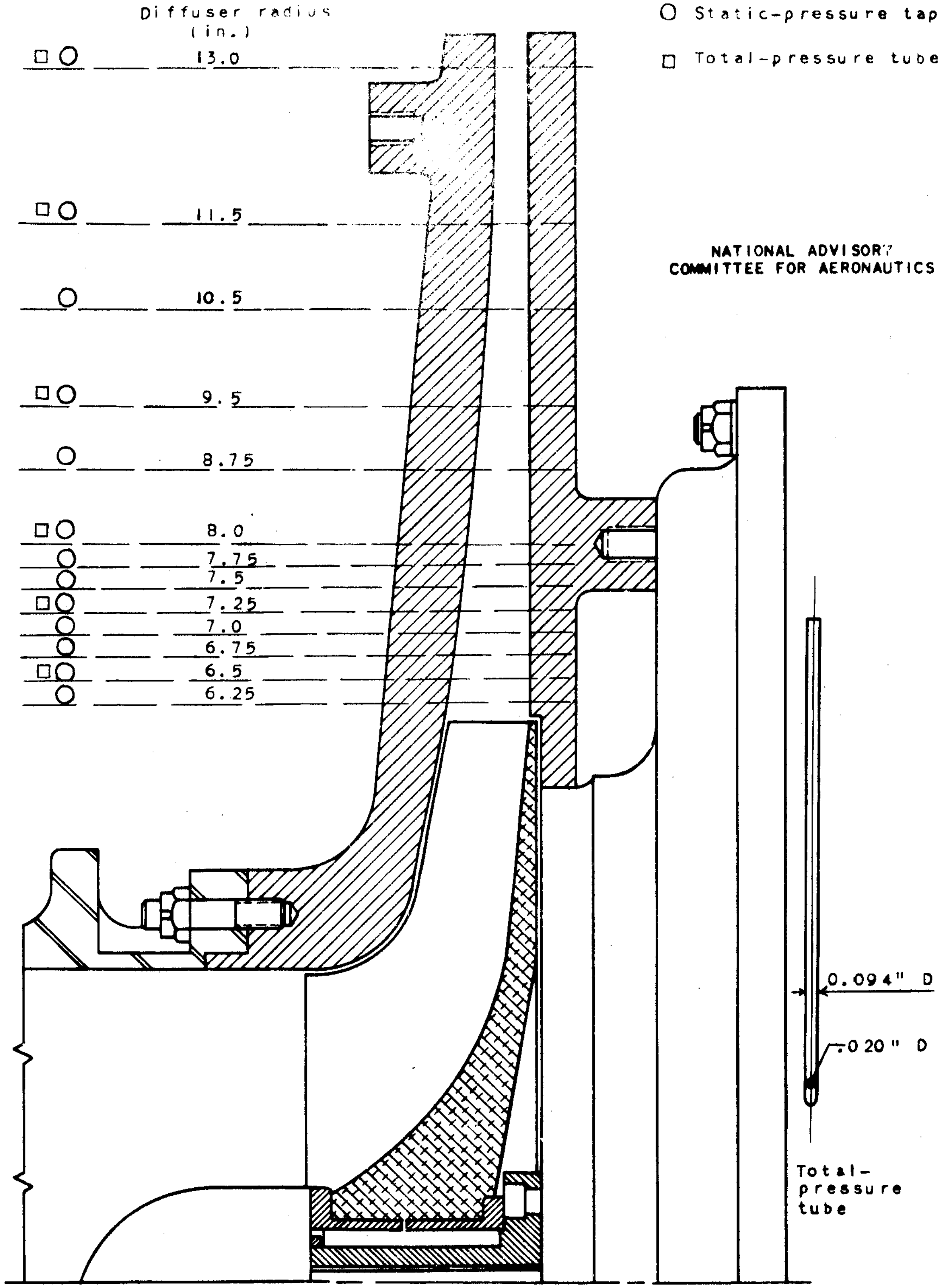

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Figure 1. - Air-passage profile in impeller and vaneless diffuser showing instrumentation along diffuser radius. Impeller radius, 6.115 inches.

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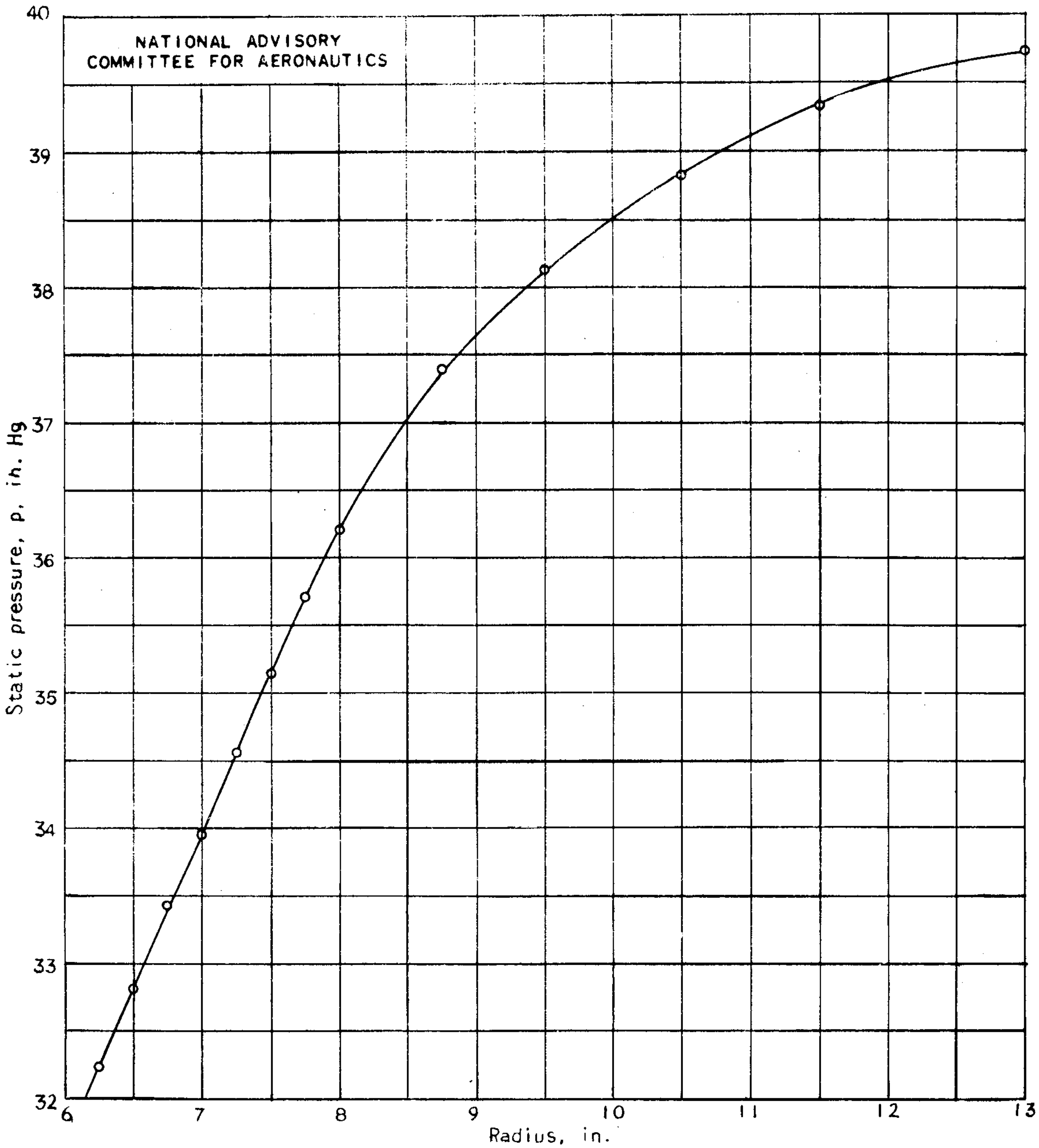

Figure 2. - Typical static-pressure distribution through constant-area vaneless diffuser.

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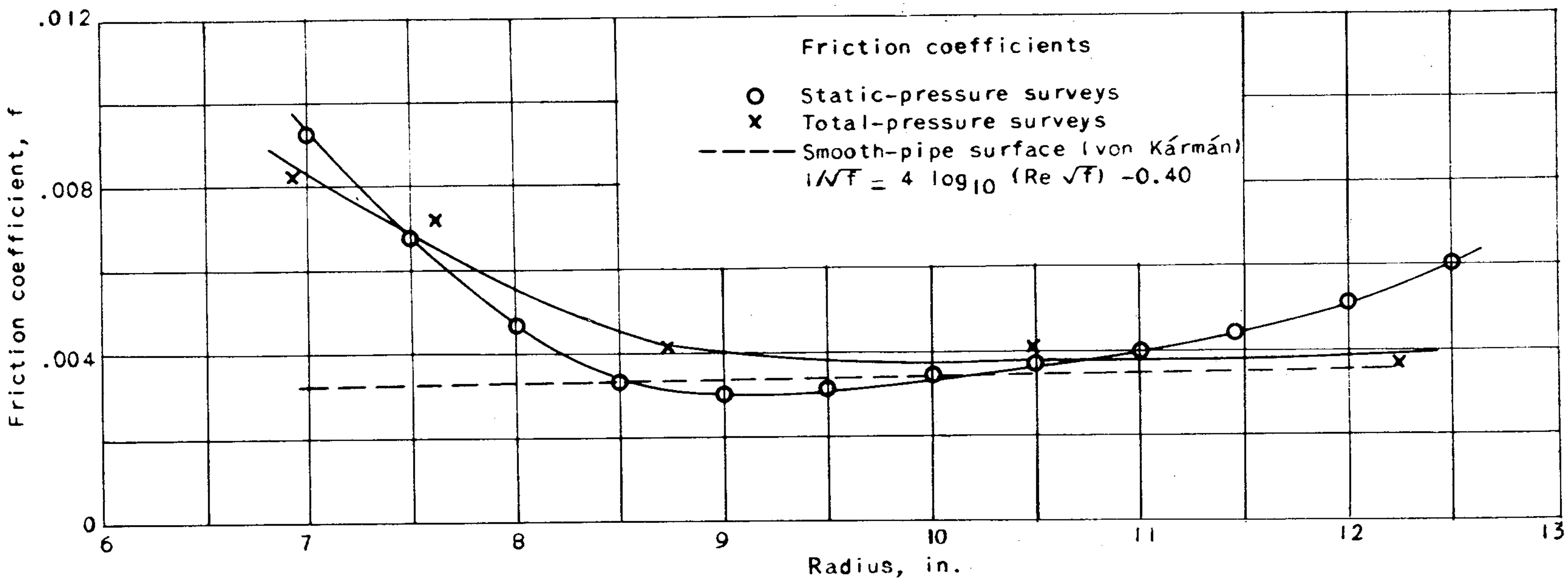

Figure 3. - Variation of friction coefficients with radius in constant-area vaneless diffuser.

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